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Proceedings of the Combustion Institute 30 (2005) 2885–2892
Proceedings of the Combustion Institute
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Combustion of lean prevaporized fuel–air mixtures mixed with hot burned gas for low-NO_x emissions over an extended range of fuel–air ratios

Naoki Aida[a], Tomoki Nishijima[a], Shigeru Hayashi[b,*], Hideshi Yamada[b], Tadashige Kawakami[a]

[a] *Faculty of Engineering, Hosei University, Japan*
[b] *Aeronautical Environment Technology Center, Japan Aerospace Exploration Agency, 7-44-1 Jindaijihigashi, Chofu-shi, Tokyo 182-8522, Japan*

Abstract

Reaction of lean to ultra-lean mixtures supported by high-temperature burned gas can resolve the dilemma between complete combustion versus ultra-low NO_x emissions in lean premixed gas turbine combustors. The combustion characteristics and NO_x emissions in "lean–lean" two-stage combustion were investigated for premixed–prevaporized kerosene–air mixtures using a co-axial flow configuration. Secondary prevaporized kerosene–air mixtures of lean to ultra-lean compositions were injected into the stream of hot burned gas prepared by the combustion of lean prevaporized kerosene–air mixtures stabilized on an annular perforated flame holder in the primary stage. The progress of mixing and reactions of the secondary mixture jets of prevaporized kerosene–air mixture injected into the co-axial primary hot burned gas flow were investigated by spatial gas sampling and direct photography. The effects of the ratio of secondary to primary air flow rates and equivalence ratios of the primary and secondary mixtures on the emissions from the two-stage combustor were studied. Imparting swirl to the secondary mixture jets resulted in an enhancement of the mixing of the jets with the primary burned gas. It was also shown that the use of reaction of lean to ultra-lean secondary mixtures supported by the hot burned gas from the primary stage is much advantageous in extending the operating range of ultra-low NO_x emissions of the 10 ppm level and complete combustion as compared with other approaches such as fuel staging and variable geometry. The proposed ultra-low NO_x combustion concept has a potential of suppressing combustion instabilities that is often experienced with lean premixed combustion.

Keywords: Low-NO_x gas turbine combustion; Flameless combustion; Lean–lean two-stage combustion

1. Introduction

It has been recognized that the precise control of the combustion zone equivalence ratio is required for lean premixed gas turbine combustor to achieve ultra-low emissions of nitric oxides, NO_x, carbon monoxide, CO, and unburned hydrocarbons, HC, over a range of engine operations. The control is conventionally made by fuel staging among multiple combustion zones and/or air flow modulation to the combustion zones [1].

* Corresponding author. Fax: +81 422 40 3445.
E-mail address: hayashi.sigeru@jaxa.jp (S. Hayashi).

1540-7489/$ - see front matter © 2004 Published by Elsevier Inc. on behalf of The Combustion Institute.
doi:10.1016/j.proci.2004.08.040

As a new approach for extending the operating range of ultra-low NO_x emissions without using complicated devices or control, a "lean–lean" two-stage combustion concept, schematically shown in Fig. 1, has been proposed. Fundamental experimental studies with premixed gaseous fuel–air mixtures have shown that the concept works as expected [2]. Rig and on-engine tests of kerosene-fueled combustors designed based on this concept have already been successfully conducted with NO_x emissions of the 10 ppm level [3,4]. An example of the operation of lean-lean two-stage combustion is as follows. The secondary fuel flow rate is modulated to increase or decrease engine power while maintaining the primary fuel flow rate constant.

This concept makes the best use of thermal reactions of very lean to ultra-lean mixtures that are injected into and mixed with hot burned gas from the lean-burn primary stage [2]. However lean, they can be completely reacted when the temperature after mixed with the primary hot burned gas is above a threshold. Good mixing of the secondary mixture with the primary burned gas is critical for successful application of this concept.

This paper describes the experimental study investigating the mixing and reactions of secondary mixture jets of prevaporized kerosene–air mixture injected into a co-axial primary hot burned gas flow. The effects of imparting swirling to the secondary jet, the ratio of secondary to primary air flow rates, and equivalence ratios of the primary and secondary mixtures on emissions are presented.

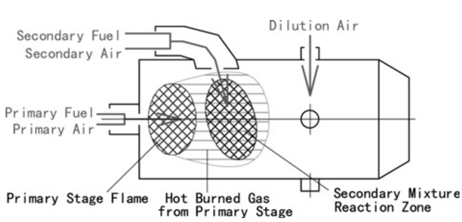

Fig. 1. Schematics of "lean–lean" two-stage combustion concept.

2. Experimental apparatus and procedures

A flow diagram of the experimental apparatus, together with a schematic drawing and a photograph of the combustor, is shown in Fig. 2. A flame tube combustor made of a transparent quartz tube, 80 mm in inner diameter, 1.5 mm in thickness, and 200 mm in length, is used for visual observation. The primary hot burned gas is prepared by burning prevaporized kerosene–air mixture on an annular ceramic honeycomb plate, 13.5-mm thick, and 80 and 27 mm in outer and inner diameters, with additionally drilled thirty 5-mm diameter holes distributed evenly over the surface. The original blockage of the honeycomb is 71%. The secondary prevaporized kerosene–air mixture is injected from a 25-mm inner diameter and 1.0 mm thick quartz tube co-axially placed in the flame tube, penetrating into the primary hot burned gas through the flame holder. The exit of the mixture injection tube is positioned 27 mm above the flame holder surface.

The secondary fuel is atomized in the mixture injection tube by means of a twin-fluid atomizer positioned co-axially in the secondary mixture injection tube. The distance from the tip of the atomizer to the exit of the injection tube is adjustable to vary the degree of prevaporization of the secondary fuel, if necessary. An axial swirler with 12 helical vanes (45° at a radius of 9.5 mm), which is removable, is placed on the body of the atomizer. The inner and outer diameters, and height of the annular passage are 14, 24, and 5 mm, respectively. The nominal swirl number is estimated as 0.67.

The primary air, supplied from a blower, is heated and then flows into the vaporizer/premixer

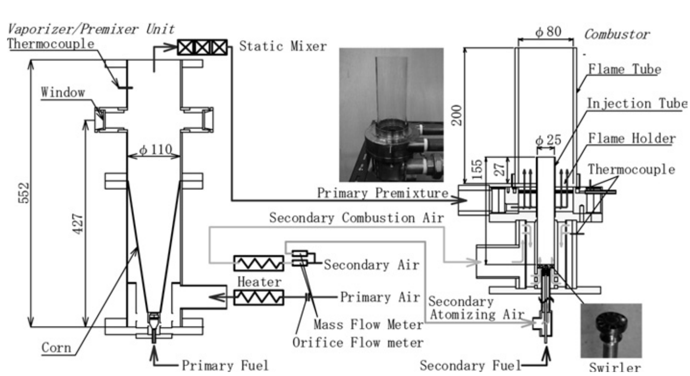

Fig. 2. Schematic drawing of experimental apparatus and photographs of combustor and swirler.

unit for primary mixture preparation. The air flow rate is measured by an orifice flowmeter before heated. The primary fuel is injected into the preheated airstream at the inlet of the vaporizer/premixer unit by using a pressure swirl atomizer with a nominal flow rate of 0.5 cm^3/s at 0.69 MPa injection pressure. A cone made of perforated metal sheet, a full angle of 15°, is co-axially positioned in the cylinder of the vaporizer/premixer unit. Air jets from the holes on the cone are provided for preventing the cylinder wall from wetting the fuel spray. A pair of glass windows are fitted to the wall of the cylinder in an opposing position. Scattering of the laser beam passed across the cylinder through the glass windows is used to check whether the fuel spray has been completely vaporized. A static mixer is placed between the exit of the vaporizer/premixer unit and the combustor inlet so that homogeneous mixtures can be supplied to the combustor.

The secondary air is supplied from a rotary compressor via a settling chamber. After electrically preheated, the majority of the secondary air enters the secondary mixture injection tube to mix with the atomized fuel. The rest, about 0.2 g/s, bypassing the air heater, is used for atomizing the secondary fuel in the twin fluid atomizer. Both airflow rates are separately measured by mass-flowmeters.

In the present experiment, the twin-fluid atomizer was axially positioned so that the tip of the atomizer was 155 mm upstream of the exit of the secondary injection tube. The temperatures of the primary mixture, Ta1, and secondary air, Ta2, were fixed at 500 K. The secondary air flow rates, Wa2, were 4.0, 6.0, and 8.0 g/s while the primary airflow rate, Wa1, was fixed at 4.0 g/s.

The fuel vaporization in the vaporizer/mixer unit and secondary mixture injection tube was confirmed to be complete. It was found in the early stage that imparting swirl to the secondary mixture jet with the swirler vanes surrounding the atomizer significantly enhanced the mixing of the secondary jets with the primary burned gas. Thus, most runs were conducted for swirling secondary mixture jets.

A single-hole gas-sampling probe attached on a two-dimensional traversing rail was used to enable spatially resolved burned gas sampling in the flame tube while an X-shaped gas-sampling rake with 32 holes in total, placed at a distance of 190 mm from the surface of the flame holder, was used to evaluate the total emissions from the combustor. The compositions of gaseous species in the sampled gas were determined by the standard gas analysis procedures: NO was measured by chemiluminescence, O_2 was measured by paramagnetic pressure difference, CO and CO_2 were measured by non-dispersed infrared absorption, and HC (as CH_2) was measured by flame ionization detection.

3. Experimental results

3.1. Progress of mixing and reaction of secondary mixture jets

The diametrical profiles of combustion efficiency and NO_x concentration were measured at different axial positions for non-swirling and swirling jets of air and mixtures. The air flow rates, Wa2 and Wa1, were fixed at 4.0 g/s. The calculated primary and secondary air velocities over the surface of the flame holder and injection tube, U1 and U2, are 1.3 and 11.7 m/s, respectively. The primary mixture equivalence ratio, $\phi 1$, was fixed at 0.68, where combustion was complete. Gas sampling was made at axial distances of 13–163 mm in 50 mm steps from the injection tube exit.

Figure 3 shows the results for non-swirling plain and swirling air jets. The combustion efficiency was calculated from the measured local CO and HC concentrations. A comparison of the NO_x concentration profiles at different axial positions for non-swirling and swirling jets reveals that imparting swirl to the jet drastically enhances the mixing of the jet with the primary burned gas. For the non-swirling jet, the development of the mixing layer is seen from the variation of the NO_x profiles with axial distance.

The secondary mixture jet, injected into and mixed with the burned gas from the primary stage, initiates reactions (combustion) in the secondary stage. In addition to the measurements of the profiles of concentration and combustion efficiency along a flame tube diameter, luminous reaction zones in the secondary stage were photographically recorded at some secondary equivalence ratios in a range of 0.6–0.1 for the primary equivalence ratio of 0.68 as above.

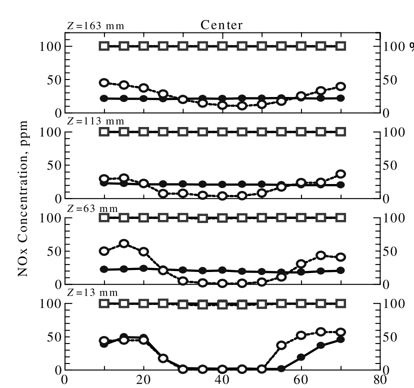

Fig. 3. Local NO_x concentration and combustion efficiency in flame tube vs. distance from wall. ($Ta1 = Ta2 = 500$ K, $Wa1 = 4.0$ g/s [$U1 = 1.3$ m/s] $\phi 1 = 0.68$, $Wa2 = 4.0$ g/s [$U2 = 11.7$ m/s] $\phi 2 = 0$, ○: NO_x □: combustion efficiency [no-swirl], ●: NO_x ■: combustion efficiency [swirl]).

A comparison of the luminous zones for non-swirling and swirling secondary mixture jets, shown in Fig. 4, shows that the swirl imparted to the secondary mixture jet has a significant influence on the developments of the secondary reaction zone. In the case of no swirl (on the left of Fig. 4), slender conical flames starting from the exit of the mixture injection tube are visible. Imparting swirl to the secondary jets makes the luminous zone much wider but shorter, as shown on the right of Fig. 4. Swirl-enhanced mixing of the secondary jet with the burned gas promotes the reactions of the secondary mixture. The luminescence of the reaction zone is decreasing in both cases as the secondary mixture becomes leaner.

The graphs on the top of Fig. 4 show the NO_x concentration and combustion efficiency profiles measured at four axial distances from the injection tube exit. Gas sampling was abandoned in the regions where HC concentration was higher than 10,000 ppm.

The plots of combustion efficiency data show that the reaction of a non-swirling secondary mixture jet occurred in the peripheral region surrounding the jet where the mixture and the primary burned gas mix. The tip of the unburned mixture volume reaches the exit of the flame tube, even when the mixture is combustible ($\phi 2 = 0.6$).

For swirling jets, ultra-lean mixtures of equivalence ratios as low as 0.1 can be completely reacted well before leaving the flame tube. "Flameless combustion" [5,6] supported by hot burned gas has proceeded with axial position. A comparison of the NO_x concentration profiles for the swirling and non-swirling secondary mixture jets shows that swirl was very effective in enhancing mixing, as before. A comparison of the combustion efficiency profiles for swirling and no-swirling secondary mixture jets shows that swirl-enhanced mixing of the secondary mixture with burned gas resulted in faster reactions of the secondary mixture.

3.2. *Two-stage combustion vs. single-stage combustion*

Typical NO_x emissions and combustion efficiency for the single- and two-stage combustion are shown for comparison in Fig. 5 as functions of overall equivalence ratio. Subscales for theoretical gas temperature and secondary mixture equivalence ratio are drawn, for reference, on the top and bottom in the figure, respectively. The fuel flow into the primary air was varied and just air was injected into the primary burned gas in the single-stage combustion. In the two-stage combustion, only the secondary fuel flow was varied while the primary fuel was fixed to keep the primary mixture equivalence ratio at 0.68.

"Parts per million corrected at 15% residual O_2 concentration" is used to present NO_x levels. Pure dilution of the primary burned gas with air does not affect the magnitude. This presentation enables us to acquire a straight comparison of the relative contributions of the primary and secondary stages to total NO_x emissions.

The steep increase in NO_x level with equivalence ratio seen for the single-stage combustion is due to an increase in the gas temperature in the primary stage. The gas temperature before dilution was estimated to vary in the 1820–2140 K range in the present runs. The well-known thermal mechanism for NO_x formation explains

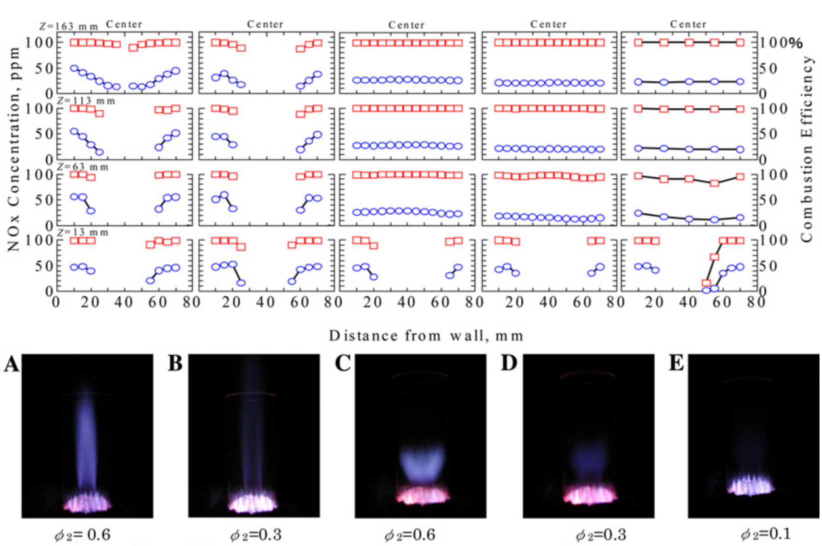

Fig. 4. Effects of swirl and secondary mixture equivalence ratio, $\phi 2$, on flame shape and diametral NO_x concentration and combustion efficiency profiles. (Ta1 = Ta2 = 500 K, $\phi 1 = 0.68$, □: combustion efficiency, ○: NO_x concentration).

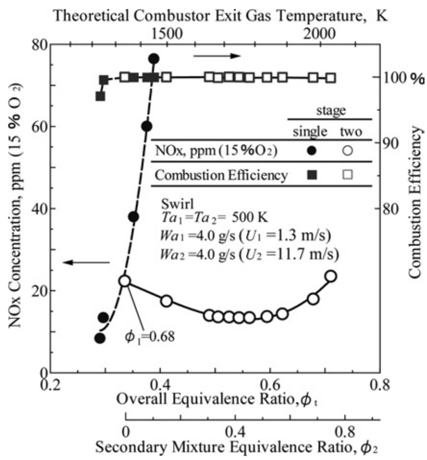

Fig. 5. NO_x emissions and combustion efficiency vs. overall and secondary mixture equivalence ratios for single-stage and two-stage combustion.

this steep increase in NO_x emissions. The combustion efficiency dropped at around $\phi t = 0.29$ (primary mixture equivalence ratio $\phi 1 = 0.58$) where the mixture was approaching the lower limit of inflammability at the mixture temperature. These observations are emissions and combustion characteristics inherent to lean premixed combustion.

In the two-stage combustion, as the secondary fuel increases, the NO_x emission level decreases from that for no secondary fuel injection ($\phi 1 = 0.68$; $\phi 2 = 0$) to a minimum at around $\phi 2 = 0.4$ and then increases. When the secondary mixture equivalence ratio reaches the fixed primary mixture equivalence ratio of 0.68, the level of overall NO_x emissions is almost equal to that for no secondary fuel injection ($\phi 1 = 0.68$ and $\phi 2 = 0$). A further increase in the secondary fuel flow rate increases the gas temperatures in the secondary stage and results in a steep increase in the NO_x emission level. The theoretical gas temperature is 1990 K at an equivalence ratio of 0.68. The combustion efficiency is more than 99.8% in the whole range of equivalence ratios tested. The "lean–lean" two-stage combustion, thus, has the potential to realize complete combustion over a wide range of equivalence ratios while maintaining very low NO emissions levels. In the single-stage combustion, an increase in the combustor exit gas temperature by 210 K (from 1230 to 1440 K) resulted in a five-time increase in NO_x emissions. In the two-stage combustion, however, very low NO_x emissions were maintained together with complete combustion, even when the combustor exit temperature increased by as much as 660 K (from 1330 to 1990 K). This feature of the "lean–lean" two-stage combustion is very attractive, since enlarging the low NO_x operating range of gas turbines is very much desired for.

3.3. Emissions at increased secondary air flow rates

The effects of the secondary airflow rate, Wa2, on the NO_x emissions and combustion efficiency are shown in Fig. 6. Wa2 was increased from 4.0 to 6.0 and 8.0 g/s whereas the primary equivalence ratio was set as close as at $\phi 1 = 0.68$ as before. A comparison of the NO_x data point for no secondary fuel at Wa2 = 4.0 g/s in Fig. 5 and the equivalent one in Fig. 6 revealed a noticeable difference (several ppm) in the baseline NO_x levels among them, suggesting that the value of $\phi 1$ for the runs whose data are shown in Fig. 6 was actually by a bit larger than that for the runs whose data are shown in Fig. 5.

As for combustion efficiency, a drop is seen at very lean secondary mixture compositions for the maximum secondary air flow rate of 8 g/s. The dent on the combustion efficiency-overall equivalence curve should be attributed to the incomplete reaction of the secondary mixtures, since the

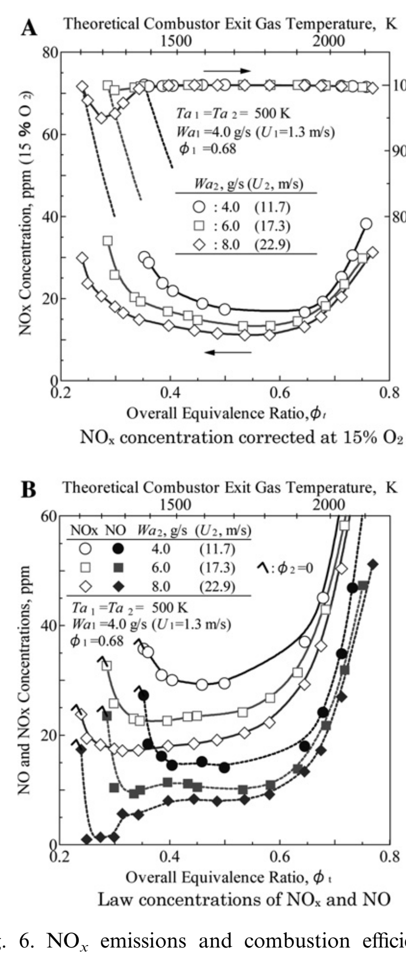

Fig. 6. NO_x emissions and combustion efficiency vs. overall equivalence ratio, ϕ_t for different secondary airflow rates, Wa2, in two-stage combustion.

combustion of the primary mixture was complete. As the secondary mixture equivalence ratio increases by increasing the secondary fuel at a given secondary air flow rate, the reaction in the secondary zone is enhanced and becomes complete before reaching the combustor exit. The dotted lines represent the combustion efficiency estimated from the measured values for no secondary fuel conditions, assuming that the secondary fuel remains unburned. A comparison of the measured data and the dotted lines shows how much the secondary fuel was reacted. The initial temperature of the secondary mixture after dilution with the hot burned gas is the most critical factor for the progress of the secondary mixture reactions, especially for ultra-lean mixture compositions.

The initial gas temperatures for $\phi 1 = 0.68$ in the second stage were approximately estimated as 1360, 1220, and 1110 K for $\mathrm{Wa2} = 4.0$, 6.0, and 8.0 g/s, respectively. It was assumed that the secondary mixture, neglecting fuel, completely mixed with the hot burned gas. It may be said that the threshold temperature for achieving complete combustion of the secondary mixture is around 1350 K.

The NO_x levels corrected at 15% O_2 for a given secondary air flow rate are once decreasing from the level for no secondary fuel and then increasing with increasing secondary fuel flow rate. The minimum NO_x level is lower at a larger secondary air flow rate. It is partly explained by a shorter residence time in the combustor and partly by quicker quenching of the NO_x formation in the primary burned gas due to enhanced mixing at an increased secondary air flow rate.

Figure 6B shows raw concentrations of NO_x and NO as functions of overall equivalence ratio. The addition of secondary fuel, while maintaining the secondary air, resulted in a reduction of both NO_x and NO. A comparison of NO_x and NO data shows a conversion of NO into NO_2. This supports the idea that the secondary fuel reduces some NO_x produced in the primary stage into species other than NO or NO_2, even at oxidizing conditions. A reduction of NO_x by hydrocarbon fuels in combustion environments at similar conditions was previously reported [2].

NO_x abatement by reburning or secondary fuel addition has been previously demonstrated at overall fuel rich or reducing conditions [7]. Miller et al. [8] recently reported that the reburning reaction occurs in locally fuel-rich diffusion flamelets at overall fuel-lean conditions. In our experiments, both primary and secondary mixtures were homogeneous and lean in contrast with their experiments. Kinetic calculations are needed to understand our observations at oxidizing conditions.

Figure 7 shows the concentrations of CO and HC emissions in the exhaust gas at the combustor exit as functions of overall equivalence ratio. At

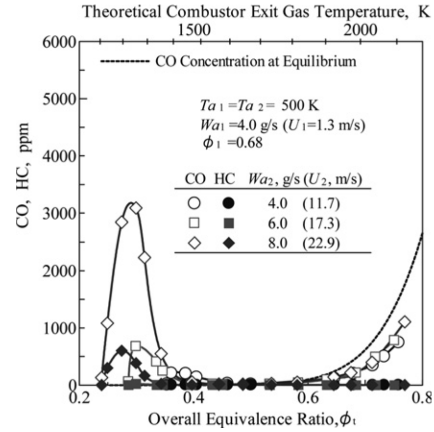

Fig. 7. CO and HC emissions and combustion efficiency vs. overall equivalence ratio, ϕ_t for different secondary airflow rates, Wa2, in two-stage combustion.

overall equivalence ratios greater than about 0.7, the CO concentrations, being less dependent on the secondary air flow rate, are increasing with overall equivalence ratio, in contrast to very low concentrations of hydrocarbons. The observed increase in CO concentration with overall equivalence ratio is consistent with the consideration based on chemical equilibrium.

On the leaner side, the CO concentrations peak at overall equivalence ratios where combustion of the secondary mixtures is incomplete due to lower temperatures of the secondary zone. The CO concentrations are much higher than the hydrocarbon concentration at the maximum secondary air flow rate of 8 g/s. Their levels decrease with increasing residence time.

3.4. Emissions for leaner primary mixtures

Emission measurements were conducted at leaner primary mixtures to further decrease NO_x emissions. $\phi 1 = 0.59$ was selected for $\mathrm{Wa2} = 4.0$ and 6.0 g/s while $\phi 1 = 0.62$ for $\mathrm{Wa2} = 8.0$ g/s to secure flame stability. The results are shown in Fig. 8. The overall combustion efficiencies remain 97–98% but the NO concentrations were of single digit in the primary zone. The temperatures of the primary stage burned gas after dilution with 4.0, 6.0, and 8.0 g/s of secondary air were calculated: 1230 and 1110 K at $\mathrm{Wa2} = 4.0$ and 6.0 g/s, respectively, for $\phi 1 = 0.59$ and 1040 K at 8.0 g/s for the $\phi 1 = 0.62$. These temperatures are by 130–70° lower, depending on the conditions, than the temperatures for $\phi 1 = 0.68$.

It is clearly seen that the reactions of very lean secondary mixtures are initiated by mixing with the hot burned gas from the primary stage, as before in Fig. 6. The dent on each combustion efficiency curve for different secondary air flow

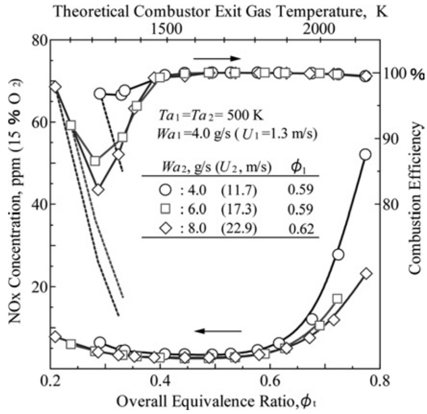

Fig. 8. Ultra-low NO_x emissions achieved by operation at lower primary mixture equivalence ratio, $\phi 1$.

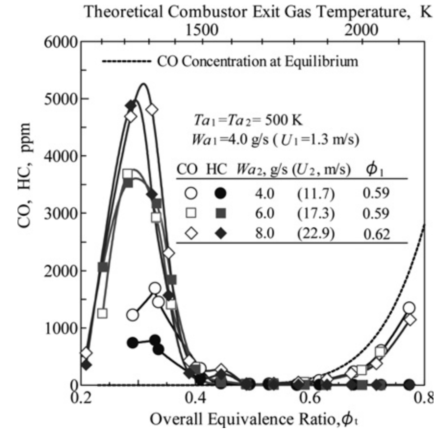

Fig. 9. CO and HC emissions vs. overall equivalence ratio at lower primary mixture equivalence ratio, $\phi 1$.

rates is, however, bigger than the counterpart in Fig. 6, where the data for $\phi 1 = 0.68$ are plotted. This is because of the lower temperatures of the primary burned gas. The effects of the secondary air flow rate on the drop in combustion efficiency at very lean secondary mixtures are consistent with those in Fig. 6. It is seen that for all secondary air flow rates, combustion becomes complete when the equivalence ratio of the secondary mixture is increased so that the final gas temperatures can be as high as about 1450 K.

A comparison of these threshold gas temperatures with previously mentioned threshold gas temperature of 1350 K for $\phi 1 = 0.68$ shows that at a higher secondary zone gas temperature, secondary mixtures can be completely reacted at lower combustor exit gas temperature. It should be noted that combustor NO_x emissions should be higher at higher primary zone gas temperature. These factors should be taken into consideration while determining the optimum air split between the primary and secondary stages.

The NO_x emissions are less than 10 ppm as long as the secondary mixture is leaner than the primary mixture and less than 5 ppm for the range $0.3 < \phi 2 < 0.6$. The behavior of NO_x emission level with overall equivalence ratio in the two-stage combustion is similar to that in Fig. 6: the NO_x level is decreasing from the value for no secondary fuel to a minimum and finally begins to increase steeply with increasing secondary fuel. It is also seen as before that the NO_x emission level approaches the level achieved for no secondary fuel when the secondary mixture is identical to the primary mixture in composition. This means that dilution of secondary combustible mixture with burned gas produced by the combustion of another mixture of the same fuel–air ratio as the secondary mixture is neutral in NO_x formation.

The NO_x levels increase steeply with an increase in overall equivalence ratio in the range $\phi_t > 0.65$ where gas temperatures are higher than 1900 K, with the order of NO_x levels for the three secondary flow rates being in the inverse order of flow rates. For $\phi_t < 0.65$, there is no practical difference in the NO_x levels for these three secondary flow rates.

Figure 9 shows the concentrations of CO and HC emissions in the exhaust gas at the combustor exit as functions of overall equivalence ratio. As far as $\phi_t > 0.4$, the dependence of the concentrations of these species on overall equivalence ratio is very similar to those in Fig. 8. This also supports the explanation based on equilibrium consideration of the burned gas.

The concentrations of HC are, in contrast to the data shown previously in Fig. 8, as high as CO concentration at $0.2 < \phi_t < 0.4$. Their levels are much higher than those in Fig. 7. A lower temperature of the burned gas resulting from the reduction in primary equivalence ratio leads to a lower secondary zone temperature. The progress of reactions at lower mixture temperatures is very slow, even if the reduction in the initial temperature due to the reduced primary mixture equivalence ratio is compensated for by increasing the equivalence ratio of the secondary mixture.

4. Conclusions

The major conclusions of the present study are as follows:

1. The use of thermal reaction of ultra-lean to lean secondary mixtures supported by the hot burned gas from the primary stage is very advantageous in extending the range of complete combustion with maintaining ultra-low NO_x emissions.

2. Imparted swirl to the secondary mixture jet significantly enhances the mixing with the burned gas from the primary stage and hence its reaction.
3. The NO_x concentration in the exhaust remains in the single-digit ppm range, corrected at 15% O_2, when the equivalence ratio of the primary mixtures is around 0.6.
4. The reaction of the secondary mixture does not result in an increase in total NO_x emissions level as long as it is leaner than the primary mixture.
5. The threshold temperature of the secondary zone for complete reaction of the secondary mixture is about 1400 K, partly depending on the primary equivalence ratio.
6. Some NO_x produced in the primary stage is reburned faster than the production of NO_x by the reaction of the secondary mixtures at exit gas temperatures lower than about 1900 K.

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Comments

Andre Nicolle, CNRS-LCSR, France. Do you have an idea concerning the respective importance of the different pathways forming NO_x at the equivalence ratio corresponding to the minimal NO_x emissions?

Reply. Our data in Fig. 6B suggests that NO_x formed in the primary stage should be reduced by fuel in the secondary mixture jet under oxygen rich conditions. Kinetic simulations of the reactions in the secondary stage are needed to explain the observed changes in the NO_x level by secondary mixture injection. Thermal NO_x mechanisms should be less important because of lower gas temperatures in the secondary stage and therefore NO formation via N_2O is probably dominant.

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Gilles Bourque, Rolls-Royce, Canada. From your emission measurements, have you calculated the air-fuel ratio? And, how does it compare with the measured air-fuel ratio?

Reply. We didn't calculate overall air-fuel ratios at each combustor cross section from the local gas compositions (Fig. 1), measured by the single-hole gas sampling probe, even when HC concentrations were measured.

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Jim Hermanson, University of Washington, USA. The use of an ultra-lean co-flow stream to impact NO_x emissions in premixed flames have been explored previously [1]. In the work cited, the configuration was somewhat different—the primary flame was stabilized by a flameholder on the combustor centerline, and there was no swirl. Still, it was shown that the NO_x emissions can be reduced, for a fixed overall equivalence ratio, in some cases by the injection of a co-flow stream below the lean flammability limit, in agreement with your results. Would you care to make any other comparative comments between the two works?

Reference

[1] J.C. Hermanson, M.B. Colket, J.J. Sangiovanni, Stability and emissions of lean, turbulent, premixed flames with a very lean co-flow, *AIAA J.* 35 (11) (1997) 1705–1711.

Reply. Although the geometry and injection of the secondary mixture of the previous study [1] are somehow different from those of the present study, the experiments in both studies have some features in common. Specifically, the interaction between the very lean secondary flow and burned gas from the combustion of the moderately lean primary flow results in lower emissions of NO_x and CO for a given overall equivalence ratio.